Analysis of vertical ground loop heat exchangers applied to buildings in the UK

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The work presented here deals with the design and performance of ground-source heat pumps and ground-sink cooling systems using vertical borehole arrays for commercial applications in the UK. Heating and cooling energy demands for a range of building and HVAC plant options are obtained by thermal modelling applied to four HVAC plant options: space heating only; heating with chilled ceilings; fan coil units and constant volume all-air plant. Ground loop designs are conducted for each system option using an impulse-response method and the parameters extracted from this are used in 10-year simulations of plant response which have been carried out using HVACSIM+. The 10-year time horizon was used to assess any degradation in earth temperature over time. The results show that a substantial reduction in energy (and, hence, carbon) can be expected of up to and exceeding 50% when using ground source heat pumps for winter heating with direct cooling in summer in association with moderate temperature cooling systems such as chilled ceilings. A degradation of earth temperature was evident with systems utilising limited cooling or no cooling but this did not appear to influence heat pump performance greatly.

Practical Applications: Design and performance data for use in vertical ground loop (borehole) heat exchanger arrays providing source heat for heat pumps as well as direct cooling for buildings are generated and reported in this paper. The data should be of help to design practitioners for the sizing of borehole arrays for both heating and cooling. Design and performance matching to a wide variety of HVAC combinations, building energy demand levels and two contrasting sets of earth thermal property data are included so that practitioners will be able to select results that suit a range of modern applications. Also included are results of 10-year energy simulations that demonstrate the required design and operating conditions needed to ensure that initial undisturbed earth conditions will not drift with time to an unacceptable extent. Comparisons are made with conventional heating and cooling methods so that estimates of carbon savings due to the use of ground-coupled heat pumps with (and without) direct cooling can be made.

1 Introduction

World capacity of geothermal energy is growing at a rate of 7.5% per year from a 2005 level of 28.3 GW.[1] Ground source heat pumps account for approximately 54% of this capacity almost all of it in north America and Europe.[1] The involvement of the UK is minimal with less than 0.04% of world capacity and yet is committed to substantial reductions in carbon emission beyond the 12.5% Kyoto obligation to be achieved by 2012. Ground source heat pumps offer a significant potential for carbon reduction and

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it is therefore expected that the market for these systems will rise sharply in the UK in the immediate years ahead given the low capacity base at present.

There are numerous ways of harnessing low-grade heat from the ground for use as a heat pump source or air conditioning sink. For small applications (residences and small commercial buildings) horizontal ground loop heat exchangers buried typically at between 1 m and 1.8 m below the surface can be used provided that a significant availability of land surrounding the building can be exploited which tends to limit these applications to rural settings. Horizontal ground loop heat exchangers can be used to circulate refrigerant (direct heat exchange) or a water/antifreeze mixture (indirect heat exchange) and rely to some extent on solar input during summer for earth temperature recovery following winter heat extraction. For a more economical use of available land, a vertical ground loop heat exchange array or "borehole" array can be used typically involving a matrix of vertical borehole heat exchangers spaced at 5 m at depths of up to 180 m. High density plastic (typically high density polyethylene) tube of 20 mm–40 mm nominal diameter is fed down a 100 mm–150 mm diameter borehole to form a U-tube with the borehole subsequently grouted using a high conductivity hard-setting compound (usually bentonite). Alternatively a double U-tube can be formed with a third and fourth pass of the pipe down the borehole. Where sufficient groundwater is available near a potential application then direct groundwater abstraction to a single heat exchanger is a third possibility which involves just two borehole details including a re-injection point for the return of the groundwater. These and other methods of low-grade ground heat harvesting have been reviewed in detail from the perspective of UK application by Rawlings.[2]

This work concerns itself with the vertical borehole heat exchange option which has greatest potential for commercial building heating and cooling possibilities in situations where direct groundwater abstraction is limited or unavailable. Sanner *et al.*[3] reviewed the growth in these systems in continental Europe together with technical and market opportunities and barriers. Three case studies involving borehole heat exchange systems for heat pump sourcing were discussed in a later work,[4] involving commercial applications with heating system capacities ranging from 22 to 330 kW with a further case study considering direct cooling at a rate of 2.5 kW from a small borehole heat exchange array. Trillat-Berdal *et al.*[5] investigated the use of a solar-assisted ground source heat pump for using two 90 m borehole heat exchangers. In this application solar panels were used primarily for heating domestic hot water discharging excess heat during periods of low or reduced hot water demand to the two borehole heat exchangers thereby raising the earth temperature for subsequent winter heat pump operation. Methods for the *in situ* measurement of earth thermal properties have been established[6] as has the performance of vertical borehole heat exchangers with alternative backfill materials for both heat sourcing and heat rejection though results were gathered over relatively short time periods only.[7] Pahud and Matthey[8] evaluated double U-tube heat exchangers and obtained significant reductions in composite borehole heat exchanger resistance over the single U-tube arrangement and Chiasson *et al.*[9] studied the impact of groundwater flows over borehole heat exchangers using modelling and concluded that the existence of moderate flows of groundwater result in conventionally-sized borehole heat exchangers being oversized for systems involving cooling-dominated loads.

Little of the previous work on these systems has considered the design requirements and performance attributes of the larger vertical borehole heat exchange systems when used for both providing a heat source for heat pumps during winter heating as well as for cooling either by means of direct "free" cooling or for condenser cooling associated with conventional air conditioning chillers. Thus the aim of the present work is to evaluate the design requirements and performance characteristics of vertical borehole heat exchange systems when applied to commercial building heating and cooling demands in typical UK conditions. Five research objectives are addressed:

- To quantify the required capacity and energy demand for both space heating and cooling for a range of UK commercial building and HVAC system attributes
- To determine design requirements for vertical ground loop heat exchange systems to meet this range of building types
- To evaluate the seasonal energy use of ground coupled systems for both space heating and air conditioning
- To evaluate the sustainability of the designs used
- To benchmark the ground-coupled solutions against conventional methods of energy supply in terms of energy use and carbon intensity

2 Energy Demand Modelling

Modelling Method

One floor of a multi-storey UK commercial building was selected as a basis for establishing a range of heating and cooling loads under various operating conditions. The building on the University of Northumbria campus in Newcastle upon Tyne is of traditional construction with cast concrete floor decks, concrete block partitions and clear double glazed windows on both major external elevations surfaces which were oriented approximately north and south (Figure 1). This exemplar was chosen for two reasons: It is of traditional commercial construction

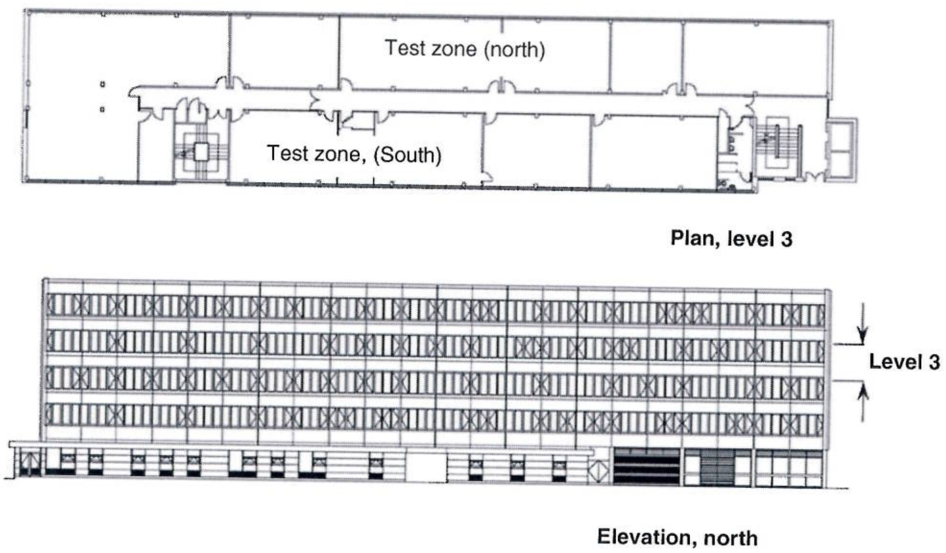

Figure 1 Northumberland Building (Top: Level 3 floor plan Bottom: North face elevation)

exhibiting many of the characteristics that are common among UK buildings of this type consisting of office type accommodation with some teaching spaces containing significant IT equipment; and certain zones of this particular building contained monitoring instrumentation enabling a comparison to be made between energy modelling results and field data. Only one typical floor (level 3) was selected for modelling. Level 3 was expressed as a 10-zone model consisting of three office-type zones along the north face, three along the south face, an un-serviced escape stairwell to the south-west, a main stairwell and lift core to the south-east, a toilet core and central corridor spine (Figure 1). The existing building is heated (using "sill-line" natural convectors) and naturally ventilated by means of openable windows.

Thermal modelling was carried out using EnergyPlus version 1.3.[10] Initially, the precise details of the existing level 3 floor plan were entered and a weather file was assembled based on data monitored at the site during 1997 shortly after the building had re-opened after extensive refurbishment in 1995. Internal temperatures and heating system control signal data that were extracted during a part of the same operating year (1997) for one typical north-facing "test zone" and one typical south-facing "test zone" (Figure 1) were also obtained. The two zones were selected because each had independent feedback control over local heat emitters and, correspondingly, internal zone temperatures, heating control signals and electrical and lighting loads had all been monitored over a period of time.

Input Data Preparation – Building Usage

The input data used for a preliminary assessment of model accuracy is detailed in the appendix identified under "Case 1: Model Verification Case". For Cases 1–3 and 5–8 usage is 07:00 h–20:00 h Monday–Friday; is 07:00 h–17:00 h Saturday; off Sunday. Occupancy was assumed to ramp in and out over a 2-hour period at the beginning and end of each occupancy cycle. Arising from survey observations a reasonable mean occupancy density of 1 person per 10 m^2 of total floor area for the existing building was applied to all modelling cases. Monitoring of lighting and small power distribution boards on all floors of the existing building resulted in an average rate of heat gain due to lighting and small power peaking at 12 $W m^{-2}$ and generally varying in proportion to occupancy. Saturday occupancy and associated casual heat gains were taken to be 50% of the corresponding Monday–Friday values. Infiltration was taken to be at a constant rate of 0.5 air changes per hour at all times. Additionally, a natural ventilation rate of 1 air change per hour (peak, and varying in proportion to occupancy level) due to openable windows was applied in all cases involving heating and natural ventilation. This value was used after a variety of test values were applied to the Case 1 model and subsequently compared with the field monitoring data.

Input Data Preparation – Plant

With the exception of Case 4 the HVAC plant was scheduled to switch on 2 hours prior to the start of occupancy. For the model test case (Case 1) the heating water flow temperature was set at 80°C with a circuit temperature differential of 10 K, consistent with existing building operating conditions. For all heat pump cases (Cases 2–8) the flow water temperature was fixed at 45°C with a circuit temperature differential of 10 K. For direct cooling options the circuit flow temperature was fixed at 16°C (generally accepted for UK applications in non-dew point air conditioning system design where surface dewing at air conditioning equipment is to be avoided) with a circuit temperature differential of 2 K. For dew-point-based plant options involving sequenced all-air plant, the chilled water flow temperature was set at 5°C with a circuit temperature differential of 5 K.

Internal zone air temperature set points were applied as follows: 21°C heating (Case 1 – to match the existing control system set up during the period in which field data were gathered); 20°C heating (all other energy modelling cases); 22°C cooling (all energy modelling cases). In all heating cases, a night set back temperature of 12°C was applied to effect frost protection.

For HVAC modelling options involving fan coil units, a fixed minimum mechanical fresh air rate of $8\,l\,s^{-1}$ per person was applied (with no additional natural ventilation) – this being a common UK design standard. For all-air sequenced air conditioning options the latter was used as a minimum mechanical fresh air input (with no additional natural ventilation) but the model set up to allow variable fresh air above this level (and up to the maximum air handling capacity of the plant) in situations where free cooling could be exploited.

The "autosize" feature in EnergyPlus[10] was used for all design zone load and plant sizing calculations and the EnergyPlus objects listed in Table A.6 in the appendix were used for plant modelling options. Primary plant options were excluded so that the overall fuel and electricity usage of the various primary options could be investigated separately outside EnergyPlus. Thus all heating and cooling energy consumptions predicted were flagged as "purchased" energy rates.[10]

Results of Model Verification

Field monitoring results for a continuous 7-week period at the end of the 1997 heating season (April/May) were extracted so that the ability of the model to predict internal conditions during heating as well as under free-float summer conditions could be evaluated. Results for this period are plotted with internal air temperatures and heating control signals predicted by the model for the two sample test zones (Figure 2). The results show that the general predictive behaviour of the model for both internal temperature and heating season activity are good. The model correctly tracks the heating system activity at the end of the heating season (weeks 1–3) and then subsequently tracks the internal temperature response driven mainly by microclimate conditions as well as user activity. The former was considered to be especially important since the objective of this section of the work was to apply the model to the accurate prediction of energy demands. When examining Figure 2 closely, the model tends to predict a more regular pattern of heating activity of moderate energy rates whereas the field data suggest shorter but larger bursts of heating demand and these differences are likely to be due to unmodelled heating and control system dynamics. However when the two sets of signal values are added over the three week period (giving, in effect, a metric which is proportional to total heating demand for the period assuming that the field signals are reasonably linear) the dimensionless results come to 27.4 (field monitoring) and 29.1 (simulated) for the north zone and 14.2 (field monitoring) and 16.0 (simulated) for the south zone. Thus overall heating system activity is predicted to be within an accuracy of 6% and 13% respectively which is considered to be good.

Energy Demand Modelling

To permit as wide a possible view of system characteristics to be matched to ground source heat pumping for winter heating as well as, possibly, ground sink cooling for summer air

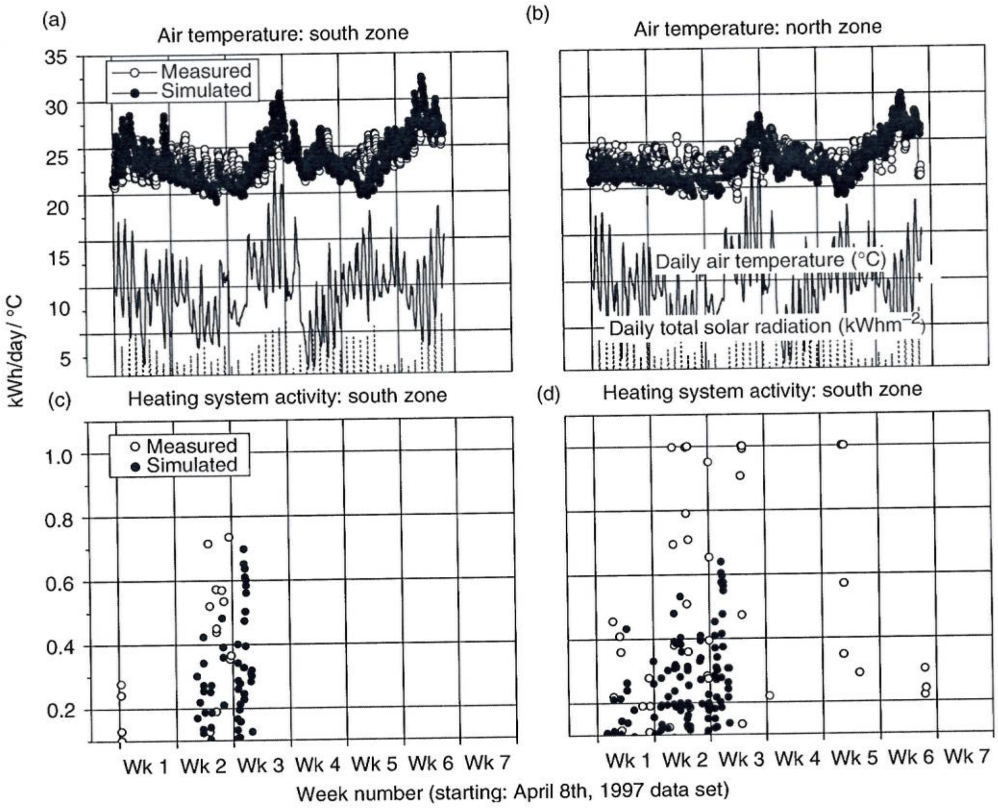

Figure 2 Results of model verification tests

conditioning, the scope of the energy modelling treatment of the test building considered above was broadened in two general ways:

- The building attributes were varied in order to explore the impact of both reduced and increased rates of energy demand with reference to the existing building
- A variety of HVAC plant options were considered including a number of cooling scenarios which go beyond the existing building set up

To deal with the first of the above departures, a range of building envelope and usage criteria were arbitrarily defined which would cast the heating and cooling energy demands at high and low extremes of what might be reasonably expected of a typical UK commercial building. The criteria applied to this range of cases are summarised in Table 1.

For the HVAC plant variants a consideration of all possibilities for UK comfort control applications would range from the simplest approach (heating with natural ventilation) through to full air conditioning. However certain air conditioning choices permit air-side free cooling whilst others are limited in this respect and the currently popular chilled surface methods such as chilled ceilings and chilled beams are usually designed to "peak lop" design cooling loads rather than fully meet them. Thus 4 HVAC

Table 1 Building envelope and usage variants

Case	Features
Base case	Existing building floor: intermediate floor of traditional construction/modest insulation standard/north-south major elevations/discontinuous occupancy
Orientation impact (traditional materials)	Existing building floor: intermediate floor of traditional construction/modest insulation standard/east-west major elevations/discontinuous occupancy
Usage impact	Existing building floor: intermediate floor of traditional construction/modest insulation standard/east-west major elevations/continuous occupancy
Insulation impact (traditional materials)	Existing building floor: intermediate floor of traditional construction/high insulation standard/east-west major elevations/discontinuous occupancy
Insulation and thermal capacity	Existing building floor: intermediate floor of low thermal capacity construction/high insulation standard/east-west major elevations/ discontinuous occupancy
Orientation impact (high insulation and low thermal capacity)	Existing building floor: intermediate floor of low thermal capacity construction/high insulation standard/north-south major elevations/ discontinuous occupancy
High energy use	Existing floor plan occupying a single storey/low insulation standard/east-west orientation/discontinuous occupancy

plant choices emerge to take all these possibilities into account:

Plant option 1 – perimeter heating with natural ventilation
Plant option 2 – perimeter heating with chilled ceilings (no free cooling)
Plant option 3 – air conditioning using 4-pipe fan coil units (restricted free-cooling)
Plant option 4 – constant volume all-air air conditioning (full free-cooling)

In the case of options 2 and 3, chilled water circulating temperatures need to be carefully selected to avoid local dewing with consequential nuisance condensation. Typically, a limiting chilled water flow temperature of 16°C is used with a narrow temperature rise (e.g. 2 K). It is therefore possible to consider a direct cooling strategy with these options in which the borehole array water is used directly in summer without the need for refrigeration. In the case of Option 4, the central all-air plant depicted would normally be designed to deliver air well below its summer dew point in order to maintain reduced air quantities and, therefore, smaller duct sizes. In turn this would require conventional chilled water at a lower temperature (5°C) but a higher temperature rise might then be possible (5 K). This option therefore requires that the heat pump undergoes cycle reversal to chiller mode in summer and the borehole array water will then be used for condenser cooling. As most of the available commercial plant tends to place a restriction on the minimum condenser water inlet temperature that can be used (in order to ensure an acceptable minimum refrigerant throttling pressure differential) the borehole array water is first passed through a heat exchanger whose secondary outlet water temperature is maintained at a minimum value. A minimum value of 20°C was used. Schematic thumbnails of each plant configuration are illustrated in Figure 3 and details of all building envelope, usage and HVAC plant variants used in the energy demand modelling can be found in the appendix.

Rather than use the locally-collected weather data (which had formed part of the model verification test) it was considered appropriate to choose a UK weather file with typical climate data characteristics suitable for energy calculations. Thus an IWEC (international weather for energy calculations) file was selected based on data collected over a number of years at Gatwick for this stage in the modelling.[10]

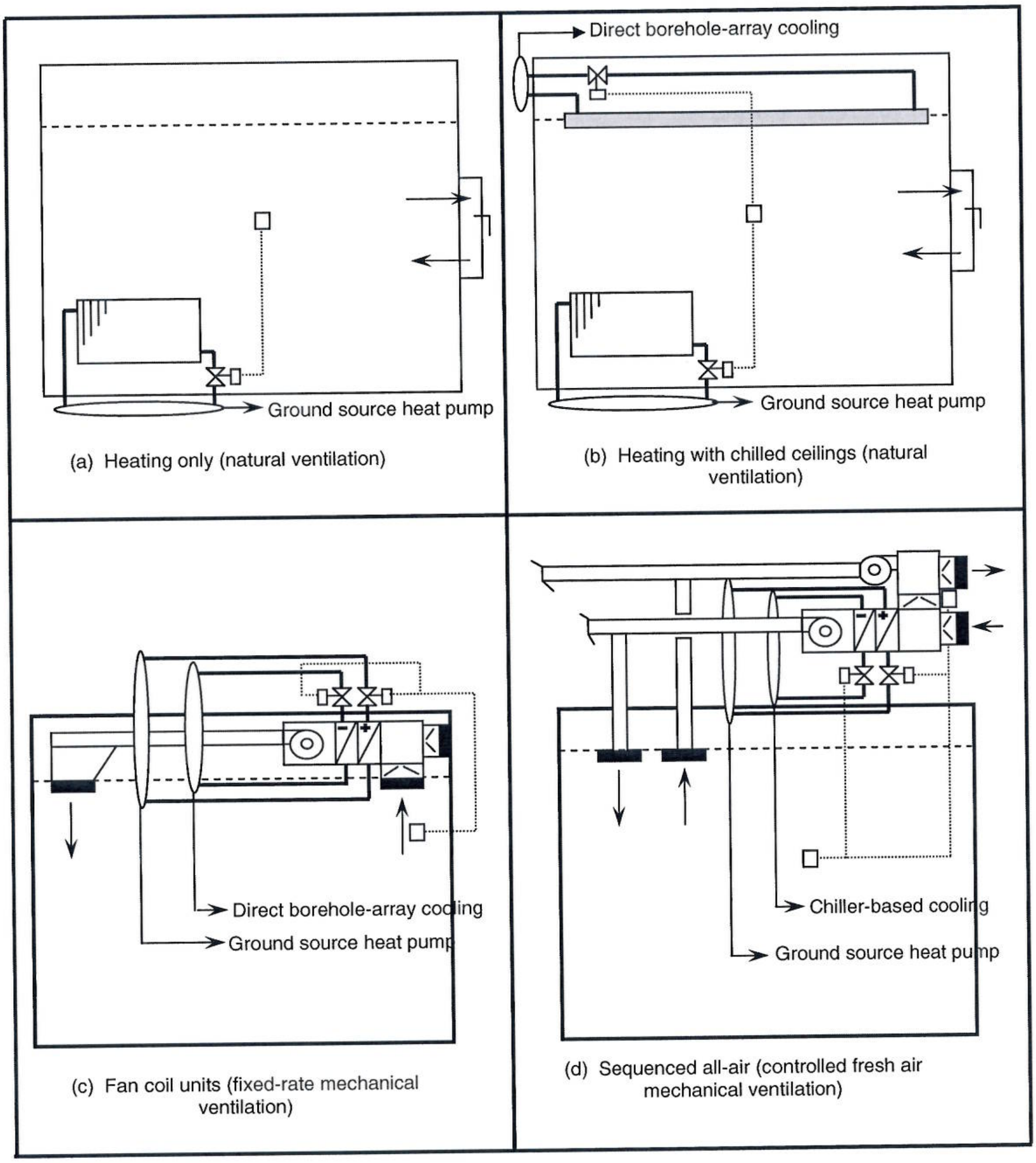

Figure 3 HVAC plant options

Results of Energy Demand Modelling Simulations

With 7 building/usage variants (Table 1) and 4 plant options a total of 28 sets of simulation results were obtained. A typical set of annual patterns of energy demands are shown for all plant options in Figure 4 (in this case for traditional construction with modest insulation with major elevations oriented east-west and discontinuous usage).

Results of annually-integrated energy demands due to heating and cooling for all cases are given in Table 2 expressed in $kWhm^{-2}$. In order to rationalise the results to a representative sample of test cases for onward treatment with the various ground-coupling possibilities three sets of results for all plant options were extracted consisting of a maximum, a minimum and a typical result. This extraction was predicated on the annual heating demand on the grounds that, for UK applications, heating will usually form the more critical comfort objective to be met over the annual cycle than cooling. The sampling was based loosely on the maximum, minimum and mean demands of the 7 results obtained from the modelling as shown in Table 2.

From this point on the resulting three energy demand cases will be referred to as

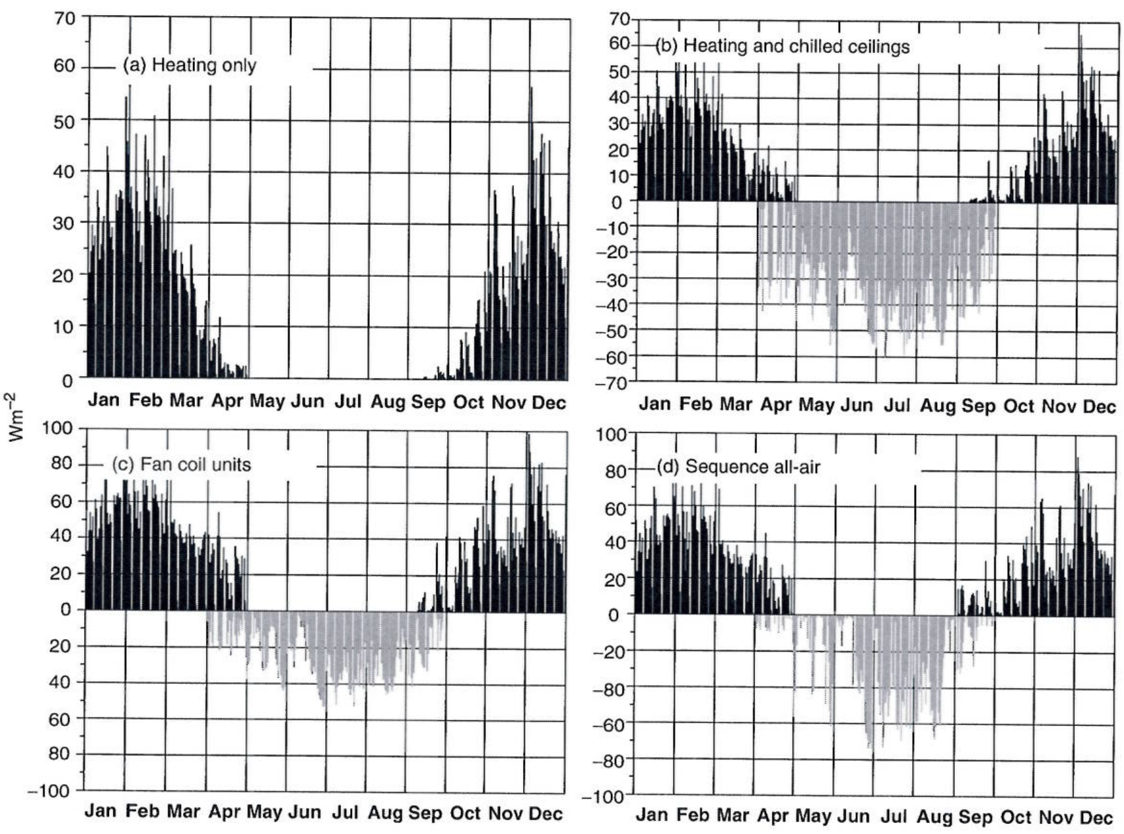

Figure 4 Typical results of energy demand modelling

Table 2 Summary energy demand modelling results

Relative Building Attributes				Annual Energy for Stated HVAC Configuration ($kWh\,m^{-2}$)							
					Heating and chilled ceilings		Fan coil units		Seqenced all-air		
Therm Cap	Insul'n	Orient'n	Usage	Heating	Htg	Clg	Htg	Clg	Htg	Clg	Designation
Trad	Typical	E-W	Typ	23.8	24.0	35.4	46.8	22.2	33.7	19.4	
Trad	**Typical**	**N-S**	**Typ**	**27.8**	**27.9**	**45.1**	**49.1**	**32.6**	**34.1**	**24.7**	**"Typical"**
Trad	Typical	N-S	Contin	28.8	29.9	71.7	39.3	53.1	29.0	39.7	
Trad	High	N-S	Typ	11.6	11.6	47.1	32.3	35.6	21.7	25.3	
Low	High	N-S	Typ	12.6	13.4	49.8	31.9	40.6	22.7	30.8	
Low	**High**	**E-W**	**Typ**	**10.5**	**11.4**	**40.1**	**31.3**	**30.4**	**22.4**	**24.7**	**"Low"**
Trad	**Low**	**N-S**	**Typ**	**86.3**	**81.0**	**30.3**	**106.4**	**23.7**	**69.7**	**26.6**	**"High"**
			Max	86.3	81.0	71.7	106.4	53.1	69.7	39.7	
			Min	10.5	11.4	30.3	31.3	22.2	21.7	19.4	
			Mean	28.8	28.5	45.7	48.2	34.0	33.4	27.3	

Table 3 Peak energy demands for the extracted sample of results

	Peak energy load for stated HVAC configuration (Wm^{-2})						
		Heating and chilled ceilings		Fan coils units		Sequenced all-air	
Relative Energy Demand	Heating	Htg	Clg	Htg	Clg	Htg	Clg
HIGH	140.5	134.0	58.8	136.2	64.3	160.1	94.8
TYPICAL	56.6	65.4	59.9	98.0	52.3	88.2	74.1
LOW	69.7	77.3	62.1	122.0	40.3	122.0	56.6

"high", "typical" and "low". Results of the peak energy demand rates (expressed in Wm^{-2}) which will be used in subsequent sections for ground loop design and heat pump selections are given in Table 3. Note that the results for "low" relative energy demand give higher peak heating demands than for "typical" relative energy demand. The reason for this is that the "low" energy demand exemplar that was selected exhibits a building with a high insulation standard but low thermal capacity (see appendix for details) whereas the "typical" energy demand exemplar used traditional materials giving higher thermal capacity and, correspondingly, greater damping to swings in energy demand than in the case of the "low" exemplar. However, as can be seen in Table 2, the overall annual energy use of the latter is considerably lower than the former.

3 Ground Loop Heat Exchanger Designs

Designs of vertical ground loop arrays meeting the energy demands arrived at in the previous section are carried out in the following. The design process used also enables modelling parameters of the ground loop arrays to be extracted for use in a detailed simulation of the ground loop arrays when coupled with heat pumps and other building heat exchange systems.

Methodology – Theory and Implementation

The vertical ground loop heat exchangers were sized with an enhanced version of the GLHEPro program.[11] The program is based on a vertical ground loop heat exchanger model[12] that is an extension of a method developed by Eskilson in 1987.[13] Eskilson's approach to the problem of determining the temperature distribution around a borehole is based on a hybrid model combining analytical and numerical solution techniques. A two-dimensional numerical calculation is made using transient finite-difference equations on a radial-axial coordinate system for a single borehole in homogeneous ground with uniform initial conditions and fixed boundary conditions predicated on an undisturbed initial earth temperature. The thermal capacitance of the individual borehole elements such as the pipe wall and grout are neglected. The temperature fields from a single borehole are superimposed in space to obtain the response from the whole borehole field.

The temperature response of the borehole field is converted into a set of non-dimensional temperature response factors called *g*-functions. The *g*-function allows the calculation of temperature change at the borehole wall in response to a step heat input for a time step. Once the response of the borehole field to a single step heat pulse is represented by a *g*-function, the response to any arbitrary heat rejection/extraction function can be determined by devolving the heat rejection/extraction into a series of step functions and superimposing the response to each step function.

Eskilson's method can be used to predict the response down to times around six hours.[13] For shorter times, the transient response of the fluid and grout can become important and an improvement to the method has been incorporated which allows a shorter time-step to be applied to account for these effects,[14] although it could not simultaneously account for variable convective resistance at the inside tube wall and the thermal mass of the working fluid. The enhanced version of GLHEPro used for this work incorporates the model developed earlier[12] that can simultaneously account for both variable convective resistance and fluid thermal mass. The enhanced model has been subject to a number of validation tests the results of which show a good agreement between the model predictions and the measured thermal response of an experimental vertical ground loop heat exchanger.[15]

The model of the ground loop heat exchanger is coupled to an equation-fit model of a heat pump so that the monthly minimum and monthly maximum heat pump entering fluid temperatures can be determined for a specific borehole configuration from the monthly total and monthly peak building heating and cooling loads. Details of the equation-fit heat pump model are given in Section 4. The sizing algorithm works by iteratively adjusting the depth of boreholes then simulating the ground loop heat exchanger for a user-selected design period. The size is adjusted until one of the user-selected heat pump entering fluid temperatures is reached for a single peak period in the simulation.

Boundary and Common Data

Results from the energy demand modelling were sorted into 12 sets of hourly time series heating and cooling energy demands each of one year duration. Each pair (heating and cooling) of time series vectors was then sorted into sets of monthly peak demands (in kW) and monthly total demands (in kWh) to form boundary conditions for GLHEPro.

To enable a direct comparison of the ground loop design requirements for all 12 energy demand and plant option possibilities a common data set was used for all cases from which the depth of each borehole heat exchanger in the array would then be allowed to vary according to each particular case. The common data used were as follows:

Array:	10 × 5 array = 50 borehole heat exchangers of varying depth, on a spacing of 5 m.
Borehole heat exchanger:	32 mm (i.d.); 34 mm (o.d.) HDPE pipe formed into a

	single U-tube; 28 mm spacing; 152 mm borehole diameter.
Borehole grout properties:	Conductivity $0.74\,\mathrm{W m^{-1} K^{-1}}$; volumetric heat capacity $3901\,\mathrm{kJ\,m^{-3}\,K^{-1}}$.
Undisturbed earth temperature:	10°C.
Design temperature limits:	Minimum heat pump entering fluid temperature between 4°C (glycol solution options) and 8°C (pure water options); maximum condenser water entering temperature (for chiller options) 20°C; maximum auxiliary heat exchanger entering temperature (for chilled ceiling and fan coil options) 14°C.

Two HVAC plant options were configured for direct borehole array cooling (chilled ceilings and fan coil units) whereas the sequenced all-air air conditioning option was arranged to make use of condenser water itself cooled from the borehole array in summer.

Case-specific Data

The earth thermal property values used were varied in order to investigate how sensitive the design (and subsequent system simulation) results might be to variations in their values. In any case earth properties tend to be highly uncertain in most cases and should be verified prior to developing a design using site investigations. Thus two sets of extreme but not atypical values for UK ground conditions were selected. In the first case an earth conductivity value of $1.4\,\mathrm{W\,m^{-1}\,K^{-1}}$ was used with a volumetric heat capacity of $1872\,\mathrm{kJ\,m^{-3}\,K^{-1}}$ – broadly equivalent to a damp sandy or sandstone earth condition. In the second case the values used were $5.2\,\mathrm{W\,m^{-1}\,K^{-1}}$ for conductivity and $2323\,\mathrm{kJ\,m^{-3}\,K^{-1}}$ for heat capacity – typical of limestone. These values were taken from the review data of Rawlings[2] and represent typical extreme values for the UK, the first set representing relatively low earth thermal diffusivity and the second set representing relatively high earth thermal diffusivity.

To assess the impact of heat pump evaporator circulating fluid temperature on both borehole array size as well as heat pump power consumption two sets of conditions for the minimum borehole array leaving fluid temperature were selected. The first consisted of fresh water with a minimum design borehole array outlet water temperature (and, hence, heat pump entering temperature) of 8°C. The second consisted of a 16% ethylene glycol solution (freezing point −5°C) with a minimum design borehole array outlet water temperature fixed at 4°C. In both cases the required circulating source fluid flow rate was determined from the peak heat extraction rates (using the peak heating loads in Table 3 with a nominal design heat pump coefficient of performance of 3.8) together with the appropriate fluid specific heat capacity and a circuit temperature differential of 4 K in all cases. Hence minimum evaporator outlet fluid temperatures of 4°C (fresh water) or 0°C (glycol solution) would be expected.

Thus 36 test cases have been evaluated: 12 energy demand and plant option possibilities each with:

- Moderate earth thermal diffusivity with fresh water as the source fluid at a minimum heat pump outlet temperature of 4°C.
- High earth thermal diffusivity with fresh water as the source fluid at a minimum heat pump outlet temperature of 4°C.

Table 4 GLHEPro design results

			HVAC Configuration							
			Htg		Htg & ch-clgs		Fan Coil Units		Sequenced All-air	
Relative Thermal Diffusivity	Relative Energy Demand	$T_{f\text{-outlet}}$ (°C)	L_b (m)	R_b (m^{2}KW^{-1})	L_b (m)	R_b (m^{2}KW^{-1})	L_b (m)	R_b (m^{2}KW^{-1})	L_b (m)	R_b (m^{2}KW^{-1})
Low	High	4	276.9	0.197	189.8	0.198	260	0.197	178.4	0.196
	Typical	4	99.8	0.255	84	0.255	97.5	0.203	86.4	0.255
	Low	4	79.1	0.255	96.5	0.255	94.4	0.199	86.5	0.199
High	High	4	149.9	0.195	118.2	0.196	145.7	0.195	125	0.194
	Typical	4	59.5	0.254	56.2	0.254	74	0.201	73.6	0.254
	Low	4	61.3	0.254	61.5	0.254	82.2	0.197	77.5	0.197
Low	High	0	121.3	0.202	81.5	0.202	116.6	0.201	78.9	0.199
	Typical	0	39.8	0.265	85.7	0.265	58.5	0.265	40.4	0.265
	Low	0	34.9	0.265	98.4	0.265	54	0.204	40.9	0.204

(Symbols used in Table 4: $T_{f\text{-outlet}}$ = evaporator source fluid outlet temperature; L_b = length of each borehole heat exchanger; R_b = overall thermal resistance of each borehole heat exchanger).

- Moderate earth thermal diffusivity with glycol solution as the source fluid at a minimum heat pump outlet temperature of 0°C.

Results – Borehole Array Designs

Besides extracting the *g*-functions needed for later system simulation, the design borehole depths and thermal resistances for each case are given in Table 4.

The variations in borehole thermal resistance evident in Table 4 are due to variations in source fluid flow rate calculated for each option based on the required peak heat transfer required and the relevant boundary conditions given in the previous section. The results confirm that the lowest borehole sizes are to be expected with systems drawing lower energy demands as well as those operating at lower circulating fluid temperatures. It also appears that those options with high cooling demands tend to result in increased borehole heat exchanger size.

4 System Simulation

To enable a critical comparison of all 36 test cases it is necessary to know what the annual energy use and carbon "cost" of each case will be. A further issue concerns thermodynamic sustainability: In heating-only cases it is expected that the initially undisturbed earth temperature will decline over time to a point where, potentially, the performance of the heat pump ceases to be viable. Equally, in situations that involve high heat rejection rates (i.e. high building cooling loads) against moderate winter heating demands, a steady elevation in earth temperature might be expected which, whilst desirable from the heat pump performance point of view, may eventually lead to a reduced capacity or indeed an incapacity to deliver free cooling in those cases that are able to exploit it. To address these issues, each of the 36 test cases was subjected to a 10-year simulation. Though most building services assets in the UK are usually written off over a 25-year maximum life span a 10-year horizon was considered sufficient to observe the required trends in performance with a reasonably manageable quantity of data. A third objective of the extended simulation was to determine practical peak rates of borehole heat transfer that might be used by the design community for the purpose of sizing and evaluating ground loop arrays in UK applications.

Methodology

The 10-year simulations were carried out using HVACSIM+[16] with an improved interface[17] due to its flexibility for multi-faceted HVAC plant simulations together with the availability of all the required component models for the application under consideration. Boundary data vectors consisting of hourly time series (one year duration) were set up for heating demands and, where applicable, cooling demands. For the 10-year simulation runs, climate data and boundary data were individually concatenated to form 10-year vectors of repeating annual data. The following Types from an extended HVACSIM+ library were adopted:

TYPE556 Heat Pump
TYPE580 Pump
TYPE620 GLHE Model
TYPE366 Ideal Heating

The TYPE556 is an equation-fit model of a vapour compression heat pump/chiller. The model used for the prediction of heat pump or chiller coefficient of performance (CoP) is as follows:

$$\mathrm{CoP} = A + BT_f + CT_f^2 + Dm_f + Em_f^2 + FT_f m_f \tag{1}$$

(in which T_f is the entering fluid temperature to the evaporator (heat pump mode) or condenser (chiller mode) in °C, m_f is the corresponding fluid flow rate in kg s^{-1} and A, B, C, D, E & F are parameters).

The parameter set can be easily obtained by fitting to manufacturers catalogue data. With the assumption of constant source and sink fluid flow rates around the evaporator and condenser, the mass flow rate terms were ignored and the parameter set based on a typical commercial reversible heat pump at the applicable capacity range given in Table 5 was used.

In heat pump mode the model assumes that the heating demand is always met. Thus the compressor load is obtained from the heating demand and Equation [1] and the evaporator load is obtained by energy balance. Similar reasoning is used in cooling mode where it is assumed that the cooling demand is always met.

The TYPE580 is a simple constant speed pump model. Estimates of source water pumping power were made by referring to typical manufacturers pressure drop data for the evaporator, plate heat exchanger (in the case of direct cooling applications), condenser (for chiller-mode applications) and the pipework network feeding a notional index borehole loop in the 5×10 array. For those cases involving low peak thermal demands the pump power used was, typically, 200 J kg^{-1} (Joules per kg of pumped source fluid) and for the larger peak heating demand cases the specific pump load increased to typically 750 J kg^{-1}. An assumed constant pump efficiency of 0.65 was used.

The modelling method which forms the basis of the TYPE620 ground loop heat exchanger is similar to that described in Section 3, and is described in fuller detail by Xu and Spitler.[12] The TYPE366 Ideal Heating model determines the fluid temperature entering the heat pump evaporator after the building cooling load has been rejected into the fluid. This component was

Table 5 Heat pump and chiller model parameters

Mode	A	B	C	D	E	F
Heat pump	3.02807	0.089012	0.000375	0	0	0
Chiller	7.029	−0.14952	0.00088	0	0	0

used only for those cases involving direct cooling.

Two alternative system strategies have been considered: The direct cooling case and the conventional "forced" cooling case. Options involving heating only used the former set-up with all cooling loads set at zero. The schematics for these alternative arrangements are given in Figure 5. A typical signal flow diagram constructed for HVACSIM+ for the direct cooling case is shown in Figure 6.

Results

Simulated year 1 and year 10 CoP results for all options are presented in Figures 7 and 8 respectively. In all cases the CoP values include the circulating borehole array fluid pump power consumption.

Results for the borehole heat rates (expressed in W/m of borehole length) are given in Figures 9 (year 1) and 10 (year 10). Note that these results express the instantaneous rates of heat transfer that occur at the instant of peak heating demand and can therefore be used as initial estimates for borehole heat exchanger sizing in UK applications for systems and building types that share similar characteristics with those considered in this work. It should however be recognised that the rates of borehole heat transfer vary substantially throughout the year below these peaks (down to a few Watts per metre) depending on the actual heating demand and prevailing source fluid

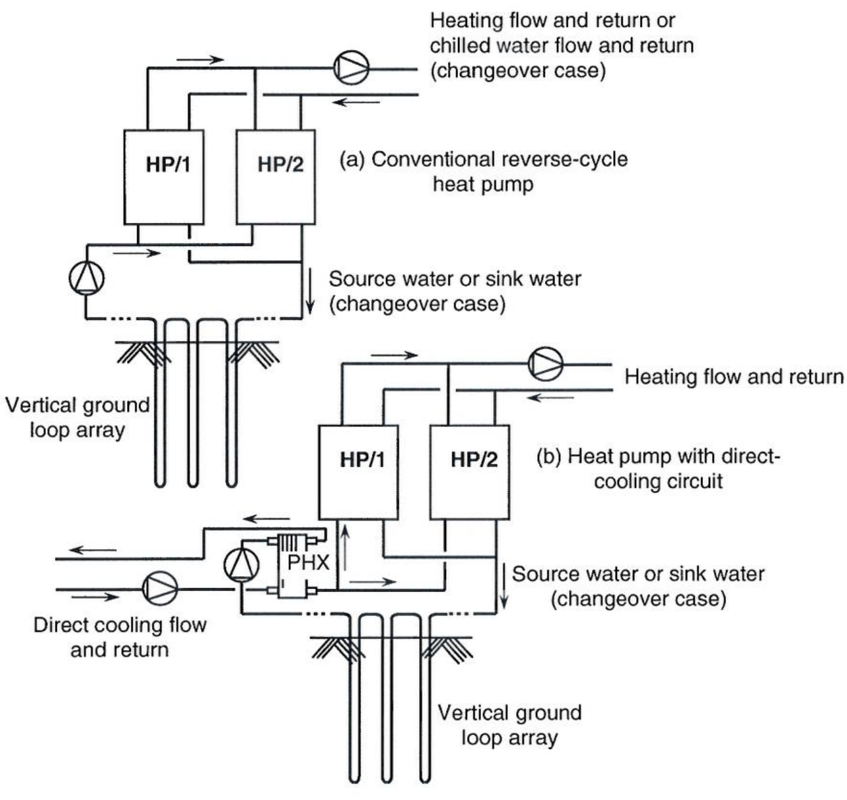

Figure 5 Primary plant connection

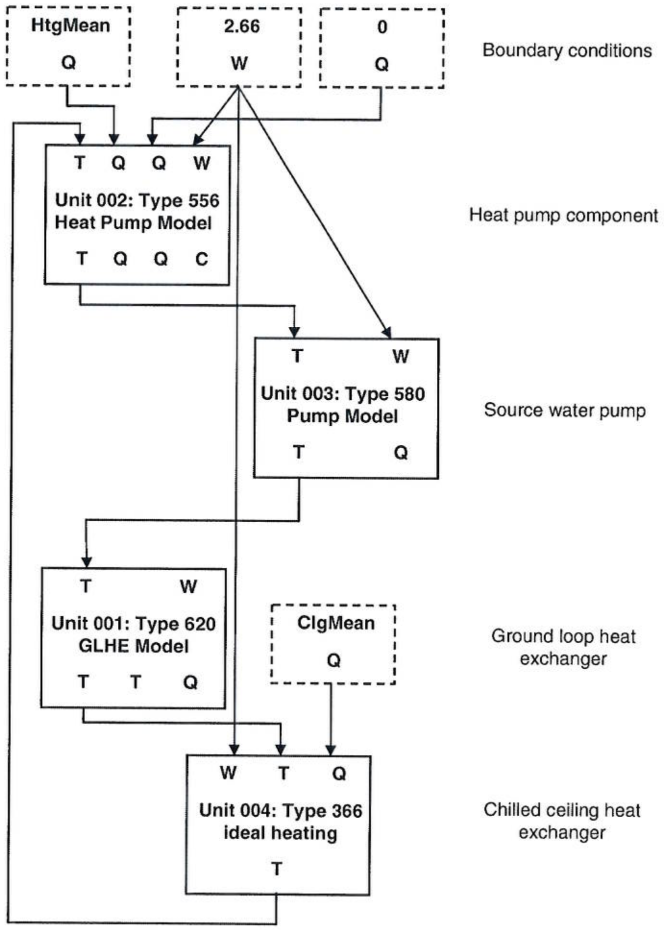

Figure 6 Typical HVACSIM+ connections

and earth temperature conditions. The rates of borehole array heat transfer expressed per metre of borehole length averaged over all variants in each HVAC plant option are given in Table 6 for both peak instantaneous conditions as well as annual average values.

A sample of 10-year simulated borehole array fluid outlet temperatures is given in Figure 11. This sample gives results for three of the plant options for high energy demand with low relatively earth thermal diffusivity and low entering fluid temperatures and for the low energy demand case with relatively low earth thermal diffusivity and relatively high entering fluid temperatures. The fan coil unit option was not included but was found to give similar results over the 10-year period to the chilled ceiling option. The results given in this set represent the most significant deviations in borehole array fluid temperature over the 10-year period of all the cases considered.

Benchmark Systems

The case for using ground source heat pumps rests entirely on the energy (and carbon) savings achievable when compared

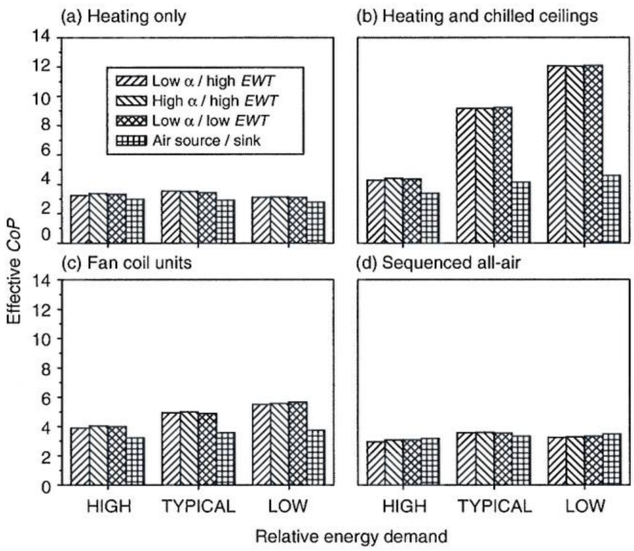

Figure 7 Plant CoP (year 1)

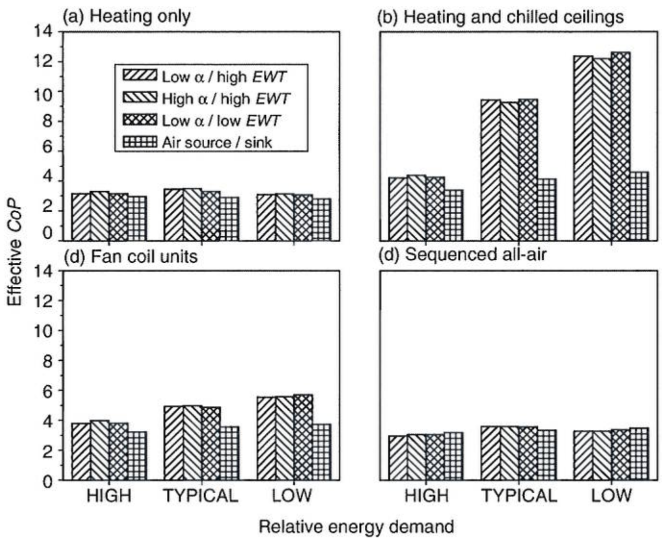

Figure 8 Plant CoP (year 10)

with a conventional approach to energy servicing. If the only choice of energy servicing were to rest with heat pumps (e.g. rural applications beyond the gas supply infrastructure) then the obvious conventional alternative would be to use an air source heat pump. For applications in which a viable gas supply is available then the most common alternative method in the UK at least would be to use a gas fired boiler which, in the context of a low-grade heating solution, would most likely

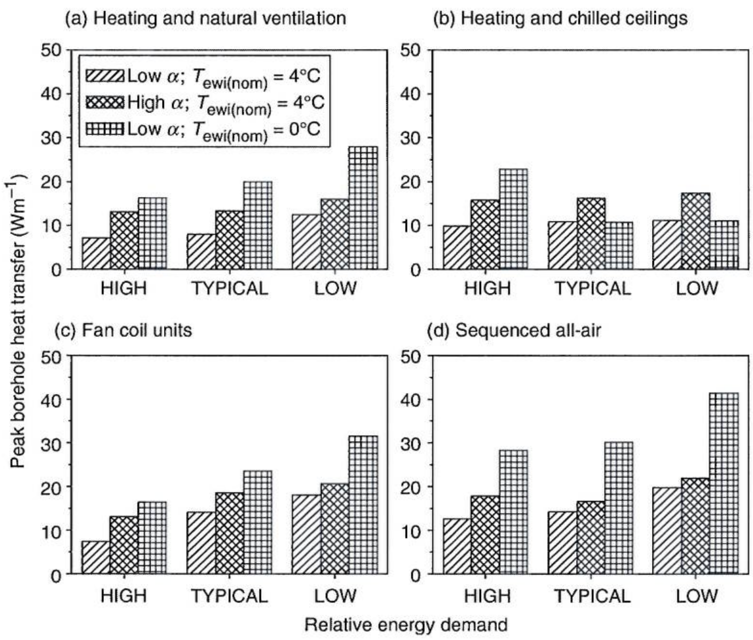

Figure 9 Borehole heat rates (year 1)

Table 6 Average rates of borehole heat transfer

HVAC Option	Peak Instantaneous Rate ($W m^{-1}$)		Seasonal Average Rate ($W m^{-1}$)	
	Heating Season	Cooling Season	Heating Season	Cooling Season
Heating only	14.9	-	2.45	-
Heating with chilled ceilings	14.0	12.8	2.17	0.96
Fan coil units	18.1	10.3	3.24	0.71
Sequenced all-air	22.5	23.2	2.42	4.88

be of the condensing type. Thus the 36 ground-source heat pump options were "benchmarked" against these two alternative system choices.

For the air source heat pump, a model similar to that described by Equation [1] was used with the exception that the entering fluid temperature would be that of ambient air. Using a commercial plant from the same basic range of equipment as used to obtain the parameters given in Table 5, parameters for an air source heat pump were obtained (see Table 7). These model parameters allow for defrosting.

A similar approach was used to predict the performance of the second benchmark test case consisting of a condensing gas boiler. Here, the fuel efficiency of typical commercial plant is known to vary with part load ratio, PLR (the part load ratio being the ratio of heat delivered at some condition divided by the heat that can be delivered at rated conditions). A reasonably representative model of boiler efficiency can be obtained using:

$$\eta = G + H \cdot \mathrm{PLR} + I \cdot \mathrm{PLR}^2 \qquad (2)$$

Table 7 Air source/sink heat pump and chiller modelling parameters

Mode	A	B	C	D	E	F
Heat pump	2.63655	0.061195	0.000365	0	0	0
Chiller	8.57795	−0.18183	0.001118	0	0	0

(in which η is the gross boiler efficiency at the given part load ratio, PLR).

A parameter set for a typical commercial condensing boiler with capacity range appropriate to the applications considered here was obtained as follows: $G=0.97270$; $H=-0.08974$; $I=0.01223$. When benchmarking against gas heating by condensing boiler, a conventional air source chiller was included for the summer cooling.

Results of the annual energy simulations for all ground source heat pump options are given in Table 8 together with the two benchmark options. These are expressed in $kWhm^{-2}$ for generality. Note that the annual energy consumptions for all ground source heat pump options are based on the average of year 1 and year 10 results (in most cases the differences between them were minor). Also given are the predicted carbon savings due to the ground source heat pump options when set against the two benchmark test cases. The carbon intensity of natural gas is $0.055\,kg\,kW\,h^{-1}$ whereas that of electricity is more problematical. A value for the carbon intensity of electricity in any economy depends on the generation mix, the efficiency of these generators and the fuels used. Furthermore the value is not static but evolves as progress is made with carbon reduction. Hence a carbon intensity chosen for today's circumstances may well be higher than the value prevailing in 10 years' time. This issue has been investigated by Bettle *et al.*[18] who recommend a value of carbon intensity for UK electricity of $0.137\,kg\,kW\,h^{-1}$ for use in present calculations. This value has therefore been used here.

5 Discussion

Overall Seasonal Coefficient of Performance (CoP)

Variations in overall seasonal CoP on account of the earth thermal property variations and design borehole fluid temperature variations considered in this work were found to be negligible. However both the building relative energy demand and the choice of plant to meet that demand were found to influence the seasonal CoP significantly. Overall mean seasonal CoP values for the heating with natural ventilation and sequenced all-air air conditioning options were almost identical at 3.3 each with standard deviations of less than 10% of the mean. However for direct cooling options using chilled ceilings or fan coil units the overall mean CoP results were 8.5 and 4.8 respectively with much wider standard deviations of 3.4 and 0.7. The relative energy demand satisfied by these two plant options was highly influential – the best results occurring when the annual cooling energy demand is high and the corresponding relative heating energy demand is low (Table 8). This is illustrated more clearly in Figure 12 which expresses the overall seasonal CoP against a per-unit floor area annual energy differential (expressed as the annual heating energy demand less the annual cooling energy demand). Plant options involving direct cooling always outperform those that either have no cooling or make use of forced (refrigeration-based) cooling but the improvement in performance is dramatic when the cooling energy demands become high. This is to be

Table 8 Energy use and carbon savings

Option	R_q	R_α	T_f	G	E_a1	E_a2	E_ghp	C_ghp	C_a2	C_a1	dC_1	dC_2
HNV	High	Low	4	91.5	0.0	29.1	27.0	3.70	3.99	5.03	7.2	26.5
HNV	Typical	Low	4	29.6	0.0	9.6	8.0	1.09	1.31	1.63	17.0	33.0
HNV	Low	Low	4	11.1	0.0	3.8	3.4	0.46	0.52	0.61	10.1	24.3
HNV	High	High	4	91.5	0.0	29.1	25.9	3.55	3.99	5.03	11.0	29.5
HNV	Typical	High	4	29.6	0.0	9.6	7.9	1.08	1.31	1.63	17.4	33.3
HNV	Low	High	4	11.1	0.0	3.8	3.4	0.46	0.52	0.61	10.8	24.9
HNV	High	Low	0	91.5	0.0	29.1	26.6	3.64	3.99	5.03	8.7	27.6
HNV	Typical	Low	0	29.6	0.0	9.6	8.3	1.13	1.31	1.63	13.6	30.3
HNV	Low	Low	0	11.1	0.0	3.8	3.4	0.47	0.52	0.61	9.2	23.6
HCC	High	Low	4	85.9	5.6	33.0	26.2	3.59	4.52	5.50	20.4	34.6
HCC	Typical	Low	4	29.6	8.1	17.7	7.9	1.08	2.43	2.74	55.6	60.6
HCC	Low	Low	4	12.1	7.2	11.3	4.2	0.58	1.54	1.65	62.5	65.0
HCC	High	High	4	85.9	5.6	33.0	25.3	3.47	4.52	5.50	23.1	36.9
HCC	Typical	High	4	29.6	8.1	17.7	7.9	1.09	2.43	2.74	55.2	60.3
HCC	Low	High	4	12.1	7.2	11.3	4.3	0.58	1.54	1.65	62.2	64.6
HCC	High	Low	0	85.9	5.6	33.0	25.9	3.54	4.52	5.50	21.5	35.5
HCC	Typical	Low	0	29.6	8.1	17.7	7.9	1.08	2.43	2.74	55.4	60.4
HCC	Low	Low	0	12.1	7.2	11.3	4.2	0.57	1.54	1.65	62.9	65.3
FCU	High	Low	4	113.5	4.5	40.4	33.9	4.64	5.53	6.86	16.1	32.4
FCU	Typical	Low	4	52.0	6.0	22.8	15.0	2.05	3.13	3.69	34.5	44.4
FCU	Low	Low	4	33.2	5.6	16.5	11.2	1.53	2.26	2.59	32.2	40.8
FCU	High	High	4	113.5	4.5	40.4	32.5	4.45	5.53	6.86	19.5	35.1
FCU	Typical	High	4	52.0	6.0	22.8	14.8	2.03	3.13	3.69	35.2	45.0
FCU	Low	High	4	33.2	5.6	16.5	11.1	1.52	2.26	2.59	32.9	41.3
FCU	High	Low	0	113.5	4.5	40.4	33.3	4.56	5.53	6.86	17.5	33.5
FCU	Typical	Low	0	52.0	6.0	22.8	15.2	2.08	3.13	3.69	33.4	43.5
FCU	Low	Low	0	33.2	5.6	16.5	10.9	1.49	2.26	2.59	33.9	42.3
SEQ	High	Low	4	73.7	6.4	30.3	32.6	4.47	4.15	4.93	−7.5	9.5
SEQ	Typical	Low	4	36.1	5.9	17.7	16.4	2.25	2.42	2.79	7.1	19.5
SEQ	Low	Low	4	23.7	5.8	13.5	14.4	1.97	1.85	2.10	−6.3	6.2
SEQ	High	High	4	73.7	6.4	30.3	31.5	4.32	4.15	4.93	−3.9	12.5
SEQ	Typical	High	4	36.1	5.9	17.7	16.4	2.24	2.42	2.79	7.5	19.7
SEQ	Low	High	4	23.7	5.8	13.5	14.3	1.97	1.85	2.10	−5.9	6.5
SEQ	High	Low	0	73.7	6.4	30.3	31.3	4.29	4.15	4.93	−3.2	13.1
SEQ	Typical	Low	0	36.1	5.9	17.7	16.6	2.27	2.42	2.79	6.4	18.8
SEQ	Low	Low	0	23.7	5.8	13.5	14.0	1.92	1.85	2.10	−3.7	8.5

Key to columns in Table 8:

Option: Plant option (HNV: heating with natural ventilation; HCC: heating and chilled ceilings; FCU: Fan coil units; SEQ: sequenced all-air air conditioning)

R_q: Relative energy demand

R_α: Relative earth thermal diffusivity

T_f: Nominal evaporator outlet fluid temperature (°C)

G: Gas boiler fuel demand (kWh per m^2 building gross floor area)

E_a1: Air source chiller electrical energy demand accompanying gas heating (kWh per m^2 of building gross floor area)

E_a2: Air source/sink heat pump/chiller electrical energy demand heating (kWh per m^2 of building gross floor area)

E_ghp: Electrical energy demand of ground source heat pump with direct or forced cooling where appropriate heating (kWh per m^2 of building gross floor area – including circulating borehole array fluid pump power)

C_ghp: Carbon "cost" of ground source heat pump with direct or forced cooling where appropriate (kg-C per m^2 of building gross floor area)

C_a2: Carbon "cost" of air source/sink heat pump/chiller (kg-C per m^2 of building gross floor area)

C_a1: Carbon "cost" of gas boiler with air source chiller (kg-C per m^2 of building gross floor area)

dC_1: Annual carbon savings (ground source/sink heat pump/chiller over air source/sink heat pump/chiller)

dC_2: Annual carbon savings (ground source/sink heat pump/chiller over gas boiler with air source chiller) expected: Cooling is delivered "free" (other than for a relatively minor pump energy "overhead") whereas the delivery of this cooling increases earth temperature resulting in an improvement in heat pump performance in the subsequent heating season.

Carbon Savings

All options delivered a carbon emission saving over a base case "benchmark" plant consisting of fully-condensing gas fired heating accompanied, where relevant, by an air cooled chiller for summer cooling.

All options delivered a carbon emission saving over the alternative "benchmark" plant consisting of air-source reversible heat pumps except for certain instances of the sequenced all-air air conditioning plant option. Specifically, when the heat extraction and heat rejection is out of balance (i.e. heat extraction dominates or heat rejection dominates) the resulting extremes in earth temperature caused the heat pump performance to deteriorate to a point where a carbon emission saving over this benchmark case could not be given. With heating-dominant loads, the earth temperature falls during the heating season causing a reduced heat pump performance but, due to the limitation of a minimum entering water temperature of 20°C to the chiller condenser during summer cooling, the plant cannot take advantage of the lower cooling temperature that is available. In addition, this option has sporadic and relatively low cooling demands because it is capable of carrying out air-side free cooling so earth temperature recovery from subsequent winter heating is minimal. When the rates of energy rejected and extracted are more in balance (which is evident in the "typical" relative energy demand applied to this option) the seasonal drift in earth temperature is reduced and a small carbon emission saving over conventional reversible air-source heat pumps is evident. Chiller-mode operation within this option will benefit once commercial equipment able to exploit very low condenser water temperatures become available.

The carbon emission saving for all options is plotted against the overall seasonal CoP in Figure 13 showing a carbon saving potential rising to in excess of 60% for cooling-dominated applications using plant options that can exploit direct cooling.

The introduction of new editions of parts L2A and L2B of the United Kingdom Building Regulations[19] in April 2006 imply a requirement to design and refurbish future buildings with a carbon saving of 28% (for typical commercial building types with some mechanical ventilation) benchmarked against the earlier 2002 standards contained within these documents. Figure 13 illustrates that this level of saving is achievable for plant operating at an overall seasonal CoP exceeding 3.2 when compared with gas-fired heating. Most of the solution options presented in this work are capable of meeting and exceeding this threshold.

Borehole Heat Transfer

With some minor exceptions, heating mode heat transfer rates required from the borehole array dominate over cooling mode heat transfer requirements (Table 6). The required rates of heat transfer (in Watts per metre of borehole) vary by approximately twofold when the minimum design fluid temperature is reduced from 4°C to 0°C at given earth thermal property values and there is negligible degradation in these rates over a 10-year term (Figures 9 and 10). The heat rates vary significantly (25–60%) at a given circulating fluid temperature between the extremes in earth thermal properties commonly found in the UK. For given values of earth thermal properties and design circulating fluid temperature, the rate of borehole heat transfer required reaches a maximum when the annual energy demand for heating and cooling are

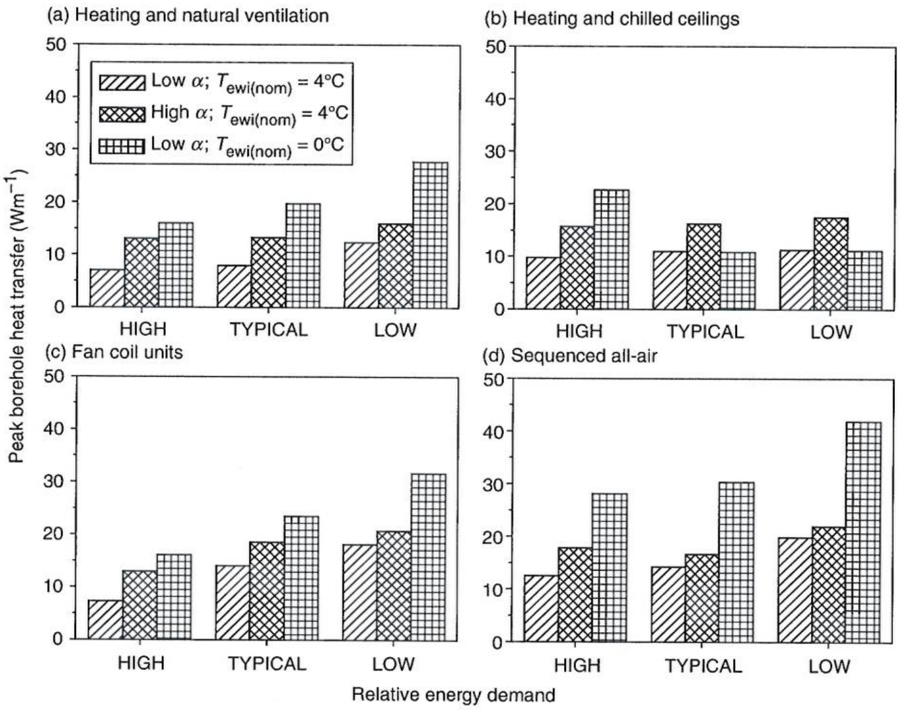

Figure 10 Borehole heat rates (year 10)

both relatively low and reasonably close to one another. For preliminary design planning, rates of heat transfer of between 15 and 22.5 W m^{-1} for a wide range of plant options and building energy characteristics would seem to be appropriate (Table 6). The use of increased borehole tube passes and high conductivity grouting materials are likely to enable higher rates of heat transfer to be used.

6 Conclusions and further work

This work has considered the design and seasonal performance of ground-coupled heating and cooling systems using vertical borehole arrays for applications in the United Kingdom. It is concluded that these systems have substantial potential for the reduction of energy (and, hence, carbon) when used in commercial buildings particularly where cooling is employed using systems that have no or little capacity to free cool using air. Compared with gas heating and conventional air-cooled refrigeration, ground-coupled systems have the potential to deliver substantial carbon savings of up to 60%. With careful design these systems are capable of meeting the new UK carbon emission reduction standard of 28% for typical commercial building types as far as energy for space heating and cooling is concerned. The best performing systems are those that reject cooling energy directly to the earth in summer and have moderate energy demands (as opposed to very low or very high energy demands) and thus have the potential to recharge earth temperatures between the heating seasons with very little power use.

This work has focused on vertical borehole arrays as the means of

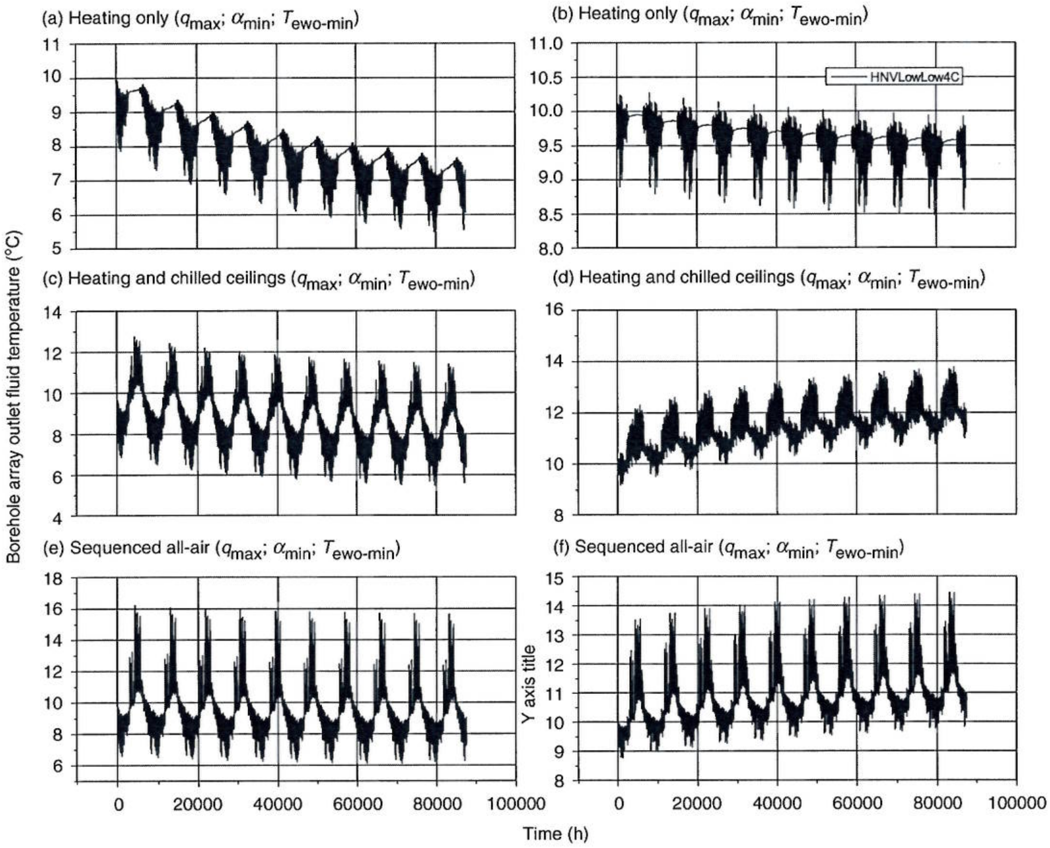

Figure 11 Sample of 10-year borehole array outlet temperatures

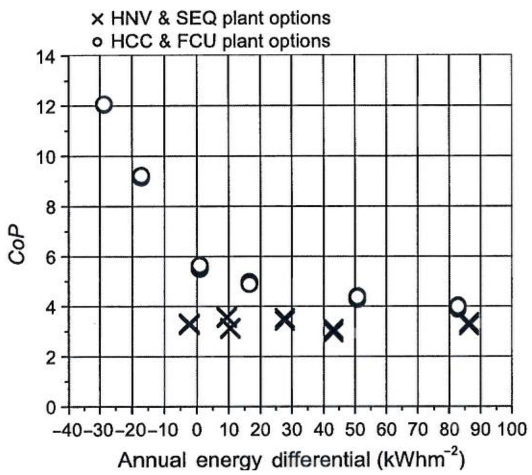

Figure 12 Overall seasonal CoP for all options

extracting/rejecting heat to earth. Further work is needed to address applications that offer direct groundwater extraction and smaller applications involving horizontal ground loop heat exchangers. More progress can also be made in reducing power usage further – for example by introducing variable speed pumping for circulating fluid through the earth loop heat exchanger. The work has used a simple curve-fit model for the heat pump which assumes flat performance across the operational range at a given source/sink temperature. More work is required to develop this model with increased rigour to enable part load performance to be accurately represented. Likewise, the benchmarking results would

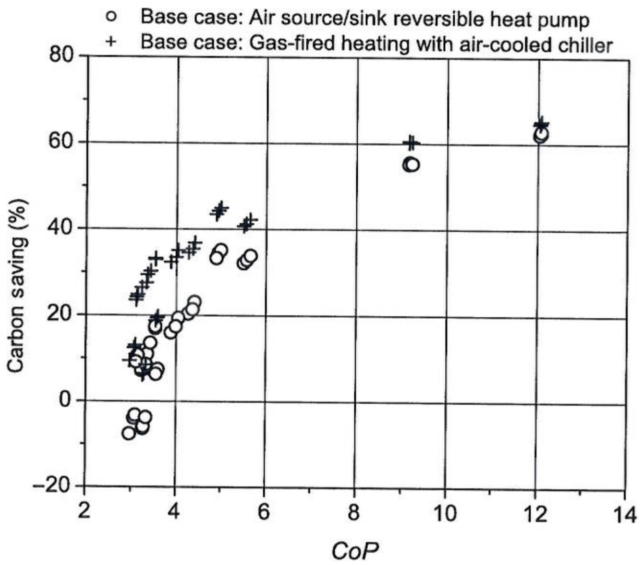

Figure 13 Carbon savings against overall seasonal CoP (all options)

benefit from a more rigorous boiler model. There is a lack of robust field data to inform UK geothermal model validation and the design of ground source heat pump systems. Further work is required on the monitoring of the thermal response of actual installations to assist model validation including a need for a national database of UK earth thermal properties.

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Appendix

Energy Demand Modelling – Input Data Summaries
Modelling cases (Table A.1).
Zone details (as modelled – Table A.2).
Construction details (cases 1, 2, 3 and 4 – Table A.3).

For Case 5.
As Table A.3 with two exceptions.

(a) Insulation thickness increased to 100 mm and this is applied to *all* external wall types.
(b) All windows become triple clear glazed (with 20 mm airspaces).

For cases 6 and 7 (Table A.4).
For Case 8 (Table A.5).
External walls and partitions as Table A.3. All windows are single clear glazed. The roof and ground floor details are given in Table A.5.

Table A.1

Case	Test	Description
CASE 1	-	Model verification case.
CASE 2	Base case	Traditional construction; typical insulation standard; north-south orientation; typical usage
CASE 3	Building orientation (traditional construction)	Traditional construction; typical insulation standard; east-west orientation; typical usage
CASE 4	Usage	Traditional construction; typical insulation standard; east-west orientation; continuous usage
CASE 5	Insulation	Traditional construction; high insulation standard; east-west orientation; typical usage
CASE 6	Thermal capacity	Low thermal capacity; high insulation standard; east-west orientation, typical usage
CASE 7	Building orientation (high insulation standard)	Low thermal capacity; high insulation standard; north-south orientation, typical usage
CASE 8	Exposure and low insulation standard	Traditional construction; low insulation standard with exposed roof and solid ground floor; east-west orientation; typical usage

Table A.2

Zone Designation	Floor Area (m^2)	Description
S1 (south, west)	81	Office-type accommodation
S2 (south, middle)	126	Office-type accommodation
S3 (south, east)	117	Office-type accommodation
N1 (north, west)	153	Office-type accommodation
N2 (north, middle)	144	Office-type accommodation
N3 (north, east)	108	Office-type accommodation
Main stairs (south, east)	42	Circulation (heated)
WC (south, east)	18	Toilet core (heated)
Escape stairs (south, west)	27	Circulation (unheated)
Corridor	102	Circulation (unheated)

Table A.3

Element	Details (outside-to-inside)
External wall (main elevations)	3 mm metallic cladding/airspace/50 mm insulation/100 mm pre-cast concrete/airspace/13 mm plasterboard and skim finish
External wall (side elevations)	112 mm facing brick/airspace/112 mm concrete block/13 mm plaster
Partitions	13 mm plaster/112 mm concrete block/13 mm plaster
Intermediate floors	13 mm plasterboard and skim/airspace/200 mm case concrete/40 mm screed/4 mm vinyl floor finish
Windows (main elevations)	Clear double glazing (20 mm airspace)
Windows (side elevations and stairwells)	Clear single glazing

Table A.4

Element	Details (outside-to-inside)
External wall (all elevations)	3 mm metallic cladding/100 mm insulation/airspace/13 mm plasterboard and skim finish
Partitions	13 mm plasterboard and skim/airspace/13 mm plasterboard and skim
Intermediate floors	13 mm plasterboard and skim/airspace (joists)/25 mm timber deck/4 mm floor finish
Windows (all elevations)	Clear triple glazing (20 mm airspaces)

Table A.5

Element	Details (outside-to-inside)
Roof	3 mm metallic cladding/100 mm pre-cast concrete planks/50 mm insulation/airspace/13 mm plasterboard and skim finish
Ground floor	25 mm rigid insulation/200 mm case concrete/40 mm screed/4 mm floor finish

Table A.6

HVAC modelling option	EnergyPlus object(s) used
Heating with natural ventilation	**BASEBOARD HEATER:Water:Convective** (zones S1, S2, S3, N1, N2, N3; main stairs; WC)
Heating with chilled ceilings	**BASEBOARD HEATER:Water:Convective** (zones S1, S2, S3, N1, N2, N3; main stairs; WC) **LOW TEMP RADIANT SYSTEM:HYDRONIC** (zones S1, S2, S3, N1, N2, N3)
Fan coil units	**BASEBOARD HEATER:Water:Convective** (zones main stairs & WC) **FAN COIL UNIT:4PIPE** (*) (zones S1, S2, S3, N1, N2, N3)
Sequenced all-air air conditioning	**BASEBOARD HEATER:Water:Convective** (zones main stairs & WC) **DIRECT AIR** (*) (zones S1, S2, S3, N1, N2, N3)

*Complete with:
COIL:Water:Cooling
COIL:Water:SimpleHeating
FAN:SIMPLE:ConstVolume.